17

INVESTIGATION OF THE EFFICIENCIES OF
A VAN WIE CENTRIFUGAL PUMP
AND
A DEMING TRIPLEX PLUNGER PUMP
BY THE
LEWIS DYNAMOMETER.

--- o ---

Chas. G. Pfeiffer.

June 10, 1901

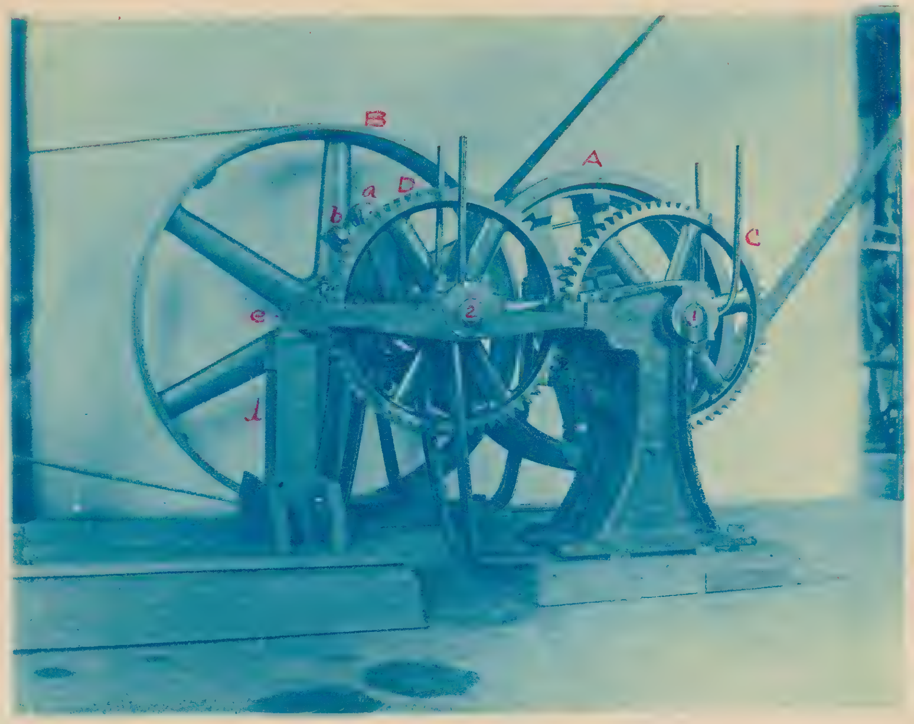
F. Victor Westermaier

20 May 1948. Lev. f. Univ. R. C. George

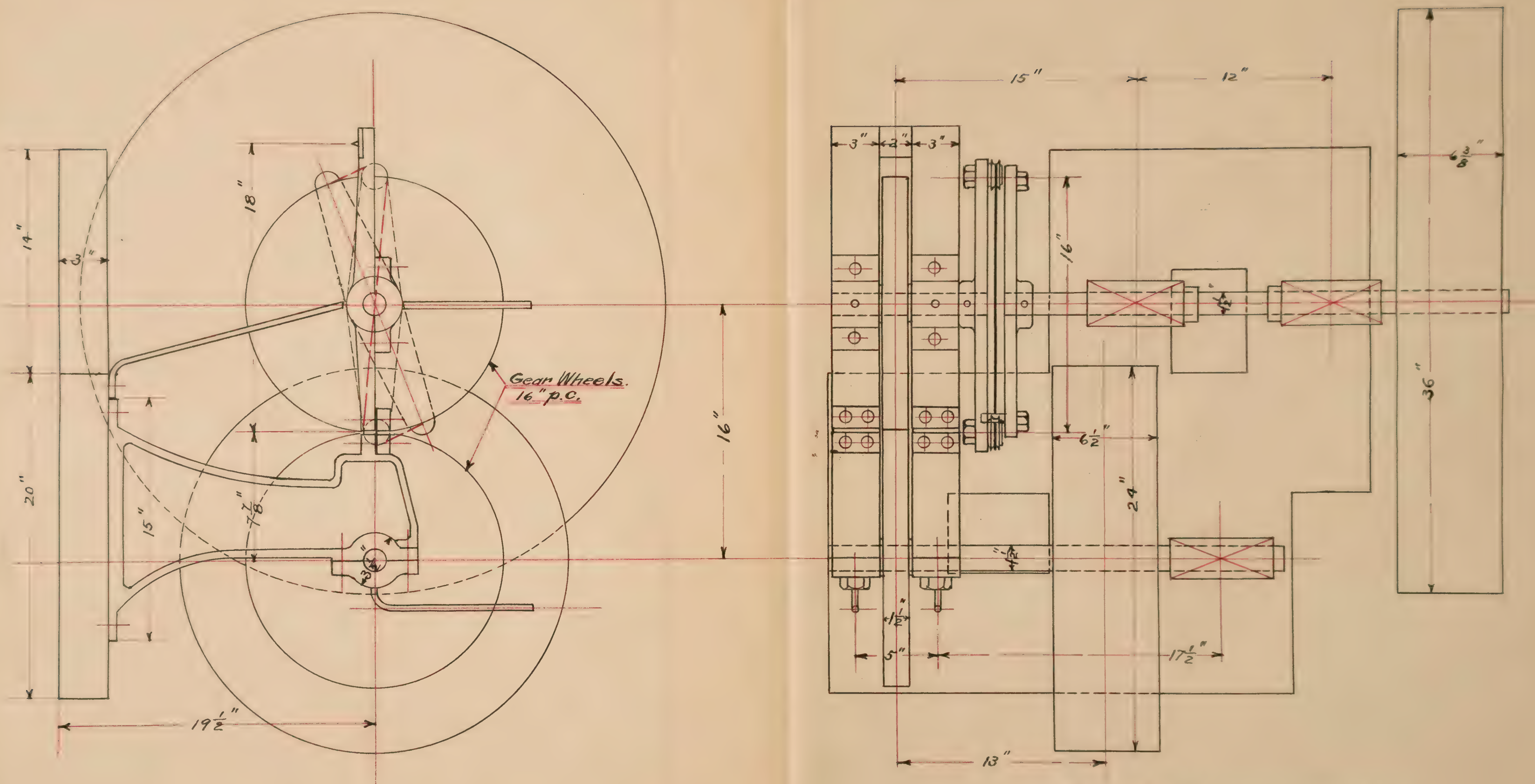
The object of this thesis was to investigate the efficiencies of a Van Wie Centrifugal Pump and a Deming 3-1/2" x 3" Triplex Plunger Pump. The power used to drive these pumps was measured by means of a Lewis Dynamometer.

THE LEWIS DYNAMOMETER.

In Mr. Lewis' discussion on "The Transmission of Power By Gearing" (Proceedings A. S. M. E. Vol. VII, Page 273) he mentions the use of this kind of dynamometer. It was necessary in his experiments to have an accurate instrument which would respond readily to changes in the delivered power.



PHOTOGRAPH. NO. I



Lewis Dynamometer.

Scale:-1½"=1 FT.

June, 3, 1901.

Photograph # 1 shows the dynamometer in the form used, and the accompanying drawing gives the general dimensions.

The pulley "A" is the driver and "B" is the driven pulley. The power is transmitted from shaft "1" to shaft "2" by the gear wheels "C", "D". The shaft of "B" is not rigidly connected to "2" but the force is transmitted through the arms "a", "b". Arm "a" has two pins on which are two hard wood rollers. A leather strap passes through the pin on "b" over the rollers on "a" and is fastened to the other pin on "b".

It is seen that gear wheel "D" is set in a frame which rests, at its outer point, on the block "d" and is supported at the other end by two very thin iron plates 3/32" thick bolted to the main frame.

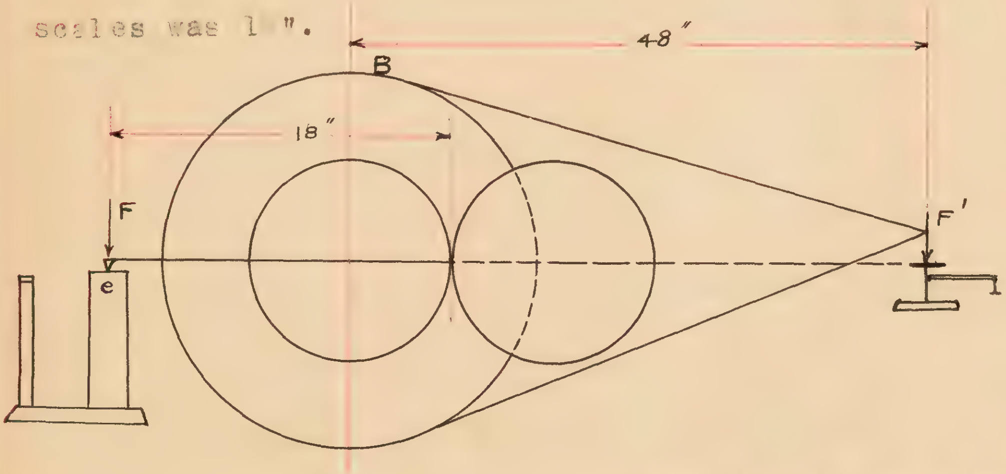
Now if there is any resistance to the turning of shaft "2" this torsional force will cause the outer end of frame "e" to be pressed down on the scale

block "d" the gear wheel "D" turning about "C" on the pitch circle. The only force, outside of gravity, which acts at the outer edge of frame "e" is due to the resistance to torsion, and any forces acting at the inner edge do not affect the conditions in the least, as the forces pass through the steel plate fulcra.

In our first experiments with the dynamometer we did not have the strap and roller arrangement but depended on two pins in the arm "a" pressing on opposite sides of "b". It was found that the vibrations in the machine were excessive and also very noisy.

A prony brake was attached to wheel "B" and loads were applied by tightening the ropes around the wheel. Readings on the dynamometer and brake scales were taken and the efficiency at different loads was calculated. The average of these was found to be 53.4 %, showing a loss of 46.6 % of the power applied, assuming the theory of the instrument to be correct. The ratio of the delivered power to the

power applied was found by the ratio of the moments of the two forces e.g. at the brake and at the dynamometer scales. The force applied at the wheel "B" is the weight of the brake plus the amount due to the tension in the ropes. The length of the arm was the distance from the center of the wheel to the knife edge of the brake. This was found to be 48". The arm of the moment at the dynamometer scales was 18".



The moment of the force applied at the wheel "B" = F ' x 48".

Moment of force applied at "e" is -

F x 18".

The efficiency is then the ratio of the

former to the latter e.g. $\dfrac{F' \times 48''}{F \times 18''} = \text{Eff.}$

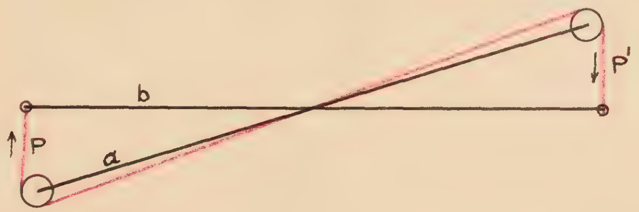
We at first made zero readings which were subtracted from the readings obtained during run. The zero reading of the brake was made by stopping the machine and taking all tension from the ropes, the wheel was turned backward and then forward, the mean of these two readings being taken. The zero reading of the scale was that reading when the machine was at rest.

In our next experiment we placed the pins on "a" on a smaller pitch circle but the efficiency remained the same as before and the racking as great as ever. This racking was caused no doubt by the fact that the pins did not exert equal pressure on the arm "b". Consequently at that point where the pressure is greatest the bending of the shaft and the journals is most, and causing the shaft to move out of line.

Our next attempt to have the pins press equally was to bore holes in ~~shaft~~ arm "b" opposite the pins

in <u>"a"</u>. The pins were then placed into these holes and rubber cushions placed at their bearings. This was as unsuccessful as the other arrangements.

At last we turned a groove on each of the pins on <u>"a"</u> and placed pins in <u>"b"</u> exactly opposite. A leather rope 3/8" in diameter was fastened to one of the pins on <u>"b"</u>, passed over the grooves in pins on <u>"a"</u> and fastened to other pins on <u>"b"</u>. The center line sketch below will explain the method.



It is seen that since the length of the rope is fixed, the pulls "P" and "P'" must be equal and hence there is only a purer twisting moment from the shaft.

TEST NO.1

DYN.SCALES	BRAKE	EFF.
124	13	.395
124	14	.493
130	16	.565
142	18	.533
149	20	.564
156	22	.588
161	24	.625

Brake zero = 9 lbs.

Dyn. zero = 89 lbs.

Speed = 320 r.p.m.

TEST NO.2

DYN.SCALES.	BRAKE	EFF.
.29.	10	.75
33	12	.75
37	14	.75
39	15	.75
41.5	16	.75

zero. 23 zero. 7 320 r.p.m

TEST NO.3

47	18	.817
52	20	.837
57	22	.851
61	24	.837

zero = 23 zero = 7 Average = 83.6

320 r.p.m.

TEST NO.4.

DYN.SCALES	BRAKE	EFF
11	10.5	.867
12	11.	.846
17	13	.875
20	14	.912
25	16	.919
29	18	.886
34	20	.896
38	22	.874
41	23	.891
44	24	.904
46	25	.896
49	26	.908
31	19	.875
26	17	.863
22	15	.892
18	13.5	.834
31	19.	.875
35	20.5	.832
20	14.	.912

zero = 3. Zero. = 7. Average = 88.26

320 r.p.m.

TEST NO.5.

DYN. SCALES.	BRAKE SCALES.	EFF.
49	15	.964
42	12	.969
44	13	.975
46	14	.937
49	15	.964
51	16	.937
53	17	.916
55	18	.900
57	19	.886
60	20	.907
62	21	.895
64	22	.887
66	23	.875
68	24	.866
zero. = 31	zero. 8	Average = .920

235 r.p.m.

TEST NO.6.

DYN. SCALES.	BRAKE SCALES.	EFF.
163	32	.975
120.5	16	.927
132	20	.927
143	24	.938
153	28	.954
163	32	.975
174	36	.906
185	40	.975
195	44	.981
174	36	.906
154	28	.954
133	20	.900
121	16	.908
143	24	.938
zero = 97.5	zero = 8	average = .939

180 r.p.m.

Test #2 shows the result of this arrangement, giving an average efficiency of 75%. This is a decided increase over the previous arrangements. We then lubricated the grooves on "a" and another test made, the results being shown in Test #3. This gives an average efficiency of 83.6%, showing that in Test #2 there had been considerable friction over the grooves, preventing the pulls "P" and P '" from being equal.

We hoped to still more increase the effciency by making the friction at the points on "a" still less. Hard wood rollers were therefore turned up and placed on the pins on "a". Test #4 shows the result of this arrangement, giving an average efficiency of 88.25%.

It was then thought that perhaps the speed also affected the workings of the machine. All these tests were run at a speed of 320 R. P. M. The speed was reduced to 235 R. P. M. by placing a smaller pulley on the line shafting driving "A".

Test #5 gives the result of this, showing

an average efficiency of 92 %.

The speed was again reduced to 180 R. P. M. Test #6 showing an average efficiency of 93.9%. It was noticed at the upper loads the efficiency was greatest often amounting to 99 %.

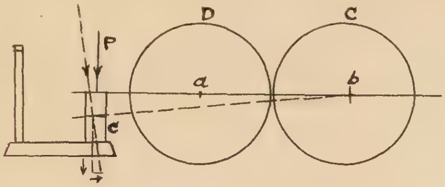
Curve #1 was plotted from two of the best tests. Its ordinates represent H. P. per revolution and the absiccas, the corrected dynamometer readings. This curve has been used in determining the power given to the centrifugal and triplex pumps when they were tested.

CONCLUSIONS.

With a tight belt, readings were easily taken at the scales, a balance being made to a half pound. It responds readily to changes in the load and different calibration tests show that for the same loads at the brake the dynamometer readings are the same. There is one thing necessary to remember in adjusting the dynamometer and that is to have the scale block at such a height that the line joining

the centers of "C" and "D" is horizontal.

In working up some of our tests we found that the efficiency came out over 100 %. By the following reasoning this can be explained:



"$\overline{ab}$" is the line joining the centers of the two wheels.

P is force acting at "e"

"c" is scale block.

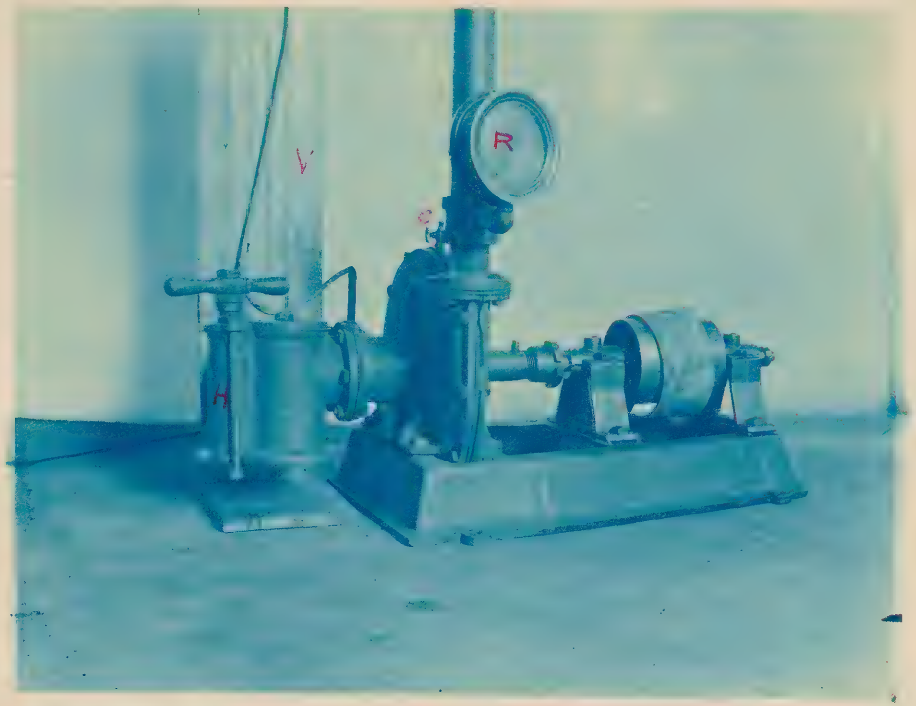
In the first position we have "$\overline{ab}$ horizontal and P acts vertically on the block "c". The scale reading in the case is accurate.

Now suppose that $\overline{ab}$ is inclined as shown, owing to "c" being shorter, the dotted line representing P is now resolved into two components, one along $\overline{ab}$ and the other vertically through the block. We measure, therefore, this amount which is only a part of the actual force, which, of course, makes our reading inaccurate.

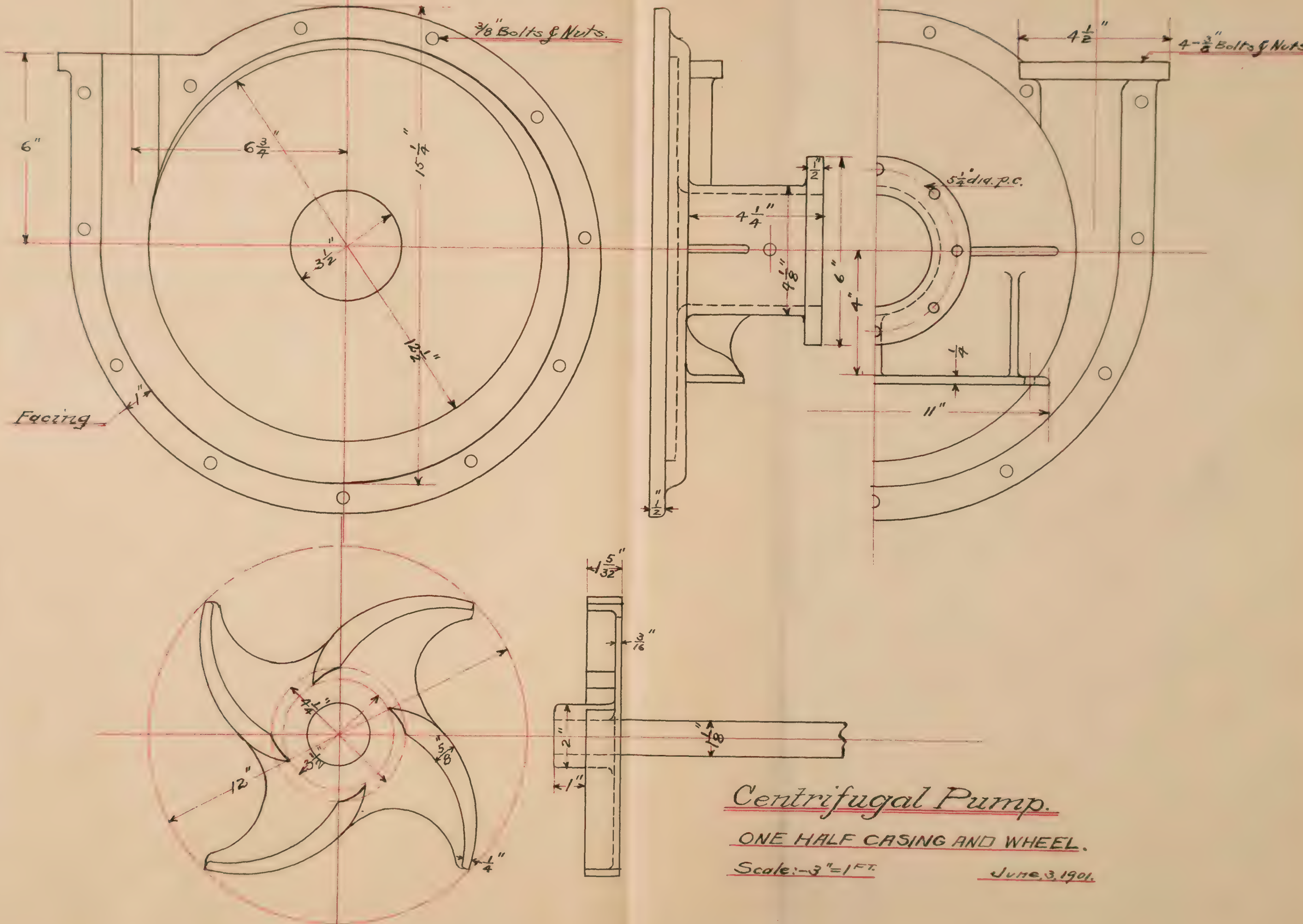
CENTRIFUGAL PUMP TEST.

The test on the Van Wie centrifugal pump was made to determine its efficiencies under different head. The photograph #2 on the following page gives a view of the pump, with the arrangement of its suction and delivery pipes, fan casing and positions of the vacuum and pressure gauges during the test. A drawing giving the principal dimensions is shown on pages 16

The power given to the pump was measured by the dynamometer, being connected to it by belt.



PHOTOGRAPH NO.2.



Centrifugal Pump.

ONE HALF CASING AND WHEEL.

Scale:-3"=1 FT.

June 3, 1901.

The pump was first started by pumping water into the casing by means of the hand pump "H", the cock "C" being opened to allow the air to escape. The power was applied and the pressure regulated by means of valve in the delivery pipe.

The suction in niches of mercury was gotten by the difference in level of the mercury in the vacuum gauge.

The pressure from pressure gauge, the R. P. M. of pump, the reading of the dynamometer, the R. P. M. of dynamometer, the reading of a hook gauge for weir in tank and the temperature of the water pumped were taken every five minutes.

The amount of water pumped per second is by Smith's formula:-

$$q = C \times \frac{2}{3} \sqrt{2g} \times b \times (h)^{3/2}$$

where "c" is a constant to be determined "b" is breadth of weir in feet and "h" is head on weir in feet.

In our experiments we found the constants for different heads by interpolating in the "Tables

For Contracted Weirs in Merriman's Hydraulics." We then plotted a curve having for its ordinates, heads on weir in feet and for abscissas, constants of the weir. This curve and its table is given on pages 38

The work done by pump in ft. lbs. is :-

$$Wk = w \times Q \times 60 \times h.$$

W is weight of a cubic foot of water at temperature considered.

Q is quantity of water over the weir per second, h is head against which water is pumped and is found by adding together the pressure expressed in feet, the vacuum expressed in feet and the distance in feet between the center of pressure gauge and the point where the vacuum was measured. This distance was found to be 1-1/2 ft.

After the tests were made, the suction pipe was taken off and the pump was run to determine its friction. This was found to be .317 H.P.

The work given to the pump from the dynamometer was gotten from the Curve #1.

The efficiency therefore is :-

$$\frac{Q \times w \times 60 \times h}{\text{Dyn. W'k}} = \text{Eff.}$$

CONCLUSIONS.

The efficiency of the pump was found to be greatest at a total head of 51 feet and the least at a total head of 44 feet. It was found that by increasing the head above 51 feet, the vacuum was lost and very litte water was pumped.

In all the tests the flow was steady.

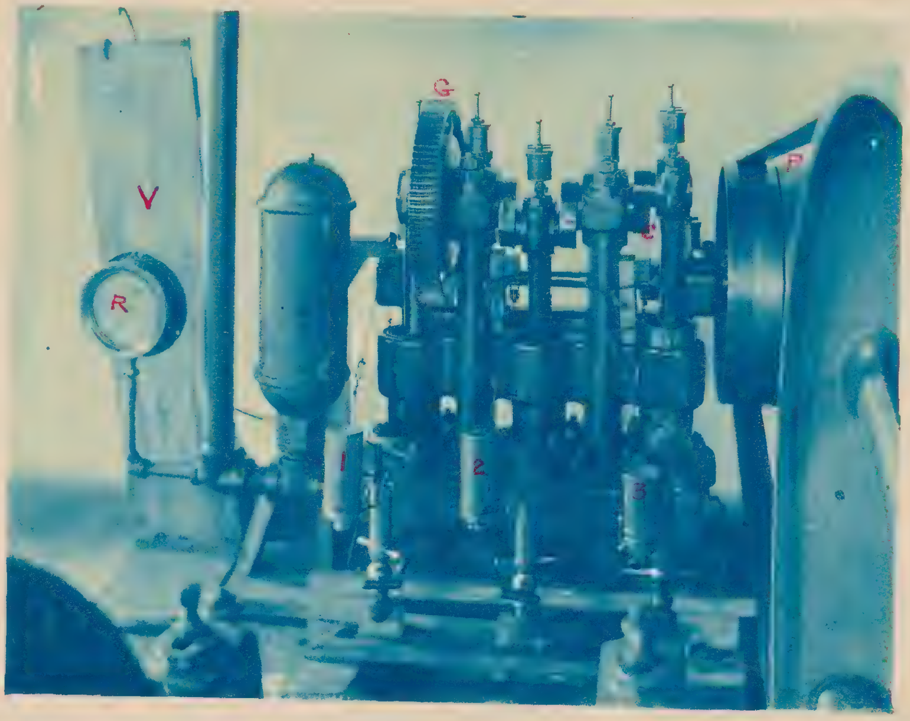
The Curve #2 shows the relation between efficiency and total head. The Table #1 gives the readings and results of the different tests.

THE TRIPLEX POWER PUMP.

In the test of this pump the efficiency was determined for different heads ranging from 14 ft. to 284 ft., the power given to the pump being measured by the dynamometer. The indicated work of the pump was found by indicating all three cylinders, the delivered work being measured by the product of the weight of water pumped and the total head through which it was pumped.

The pump consists of three hollow plungers 3.49" in diameter and having a stroke of 3". The cranks are set 120° apart.

Taking the photograph on the next page and by following the letters on it, the workings and arrangements of the parts of the machine can be understood.



PHOTOGRAPH NO.3.

The power is transmitted by the fast pulley "P" to the counter shaft on which there is a small pinion gearing into the large gear wheel "G" which is fastened to the crank shaft "C".

The suction is through a 2" pipe in the back of the pump and the delivery is 1 1/2". Each of these cylinders takes its suction from a common box and delivers into the delivery box. The indicators were placed as shown in photograph. In order to get straight line motion for the indicators, brass screw eyes were fastened into the tops of the plungers and strings fastened to these, were passed over pulleys to the indicators.

The revolutions were counted by bolting a rod to the connecting rod of cylinder #1, and connecting this by a string to a revolution counter in the back of the machine.

The suction was measured by the vacuum gauge "V" and the pressure, by the gauge "R". The suction and pressure were reduced to feet and to

this was added the distance in feet between the center of the pressure gauge and the point on the suction pipe at which the suction was measured.

The water pumped was measured by passing over a weir. As the head on the weir .08 ft. was so small the weir had to be calibrated by running water over the weir at that head and getting the amount per second by using the hook gauge in the cistern and the calibration curve of the cistern.

The mechanical efficiency of the pump is the ratio of the Indicated to the Dynamometer work, or :-

$$\frac{\text{I. H. P.}}{\text{D. H. P.}} = \text{Eff.}$$

The efficiency of the pump itself is the ratio of the delivered work, measured by the product of the weight of water pumped and the height of total head, and the indicated work, or :-

$$\frac{\text{D. H. P.}}{\text{I. H. P.}} = \text{Eff.}$$

A test was made to determine the slip or leakage past the valves. The total amount of water displaced by the plungers is equal to the products of the areas of the plungers and the stroke by the number of strokes made per minute. This was found to be :-

Area = .10.964 "

Stroke = 3"

$$\frac{10.964 \times 3 \times 80 \times 3}{1728} = 4.568$$

Cu. Ft. per minute.

No. of Strokes = 80

Water actually pumped per minute was found from calibration curve of cistern to be 4.20 cubic feet. Therefore the loss or slip is .368 cu. ft. per minute.

The per centage slip is, therefore,

$$\frac{.368}{4.568} \times 100 = 8.05\ \%$$

The indicator springs were tested on the steam gauge tester. Cards were taken at the different known pressures and the scale of the springs thereby found. The following corrections were made:-

50 # (Tabor)	-	[illegible]2.3 #	[illegible]	in	Cyl. #2
40 # (Tabor)	-	42.[illegible] #	"	"	Cyl. #2
40 # (Thomson)	-	43.	"	"	Cyl. #3
100 # (Thomson)	-	100 #	"	"	Cyl. #3

The pressure gauge was also calibrated and a curve plotted between correct and gauge readings. The calibration curve and tables are given on pages 39-40

Tables showing the M. E. P. and indicated work in each of the cylinders are given on pages

A test was also made to determine the friction in the different parts of the machine. It took to run:-

Counter shaft and belt	--	.058	H. P.
Crank Shaft,	--	.065	H. P.
Plunger #1	--	.201	H. P.
" #2	--	.180	H. P.
" #3	--	.144	H. P.

Total friction of pump was, therefore, .648 H. P.

The amount of work to run pump altogether was

equal to .648 H. P.

CONCLUSIONS.

The pump delivers a constant quantity of water under all pressures. Curve #3, page shows the relation between pump efficiency and total heads pumped against.

The card tables show that the M. E. P. is not the same in all the cylinders, due to irregularities in structure.

It is seen that the losses due to friction are considerable, but are due to the fact that the pump was tested when new and the parts had not time to settle down to their bearings.

It is also seen from the table of efficiencies of the pump that the ratio of the delivered work to the indicated was about constant, being about 74 %.

Throughout the test there was no leakage about any of the plungers; showing good costruction.

Tables, Curves
and Cards.

CENTRIFUGAL TEST.

Test No.	Suction Inches.	Suction Ft.	Pressure	Correct Pressure	Pressure in feet.	Total Head.	Head on Weir. In.	Head on Weir. Ft.	Temp.	Time.
1	11.45	12.927	25	16.8	38.707	53.134	3.10	.258	85	10.55
2	14.20	16.031	23	14.6	33.638	51.169	3.43	.286	84	11.05
3	15.50	17.499	21	12.5	28.800	47.799	3.50	.292	85	11.40
4	9.55	10.782	27	18.8	43.315	55.597	2.84	.237	84	11.50
5	9.10	10.274	28	20.0	46.080	56.354	2.79	.232	85	12.05
6	11.40	12.871	26	17.8	41.011	53.882	3.12	.260	86	12.15
7	13.40	15.129	24	15.6	35.942	51.071	3.35	.280	86	12.23
8	14.00	15.806	22	13.5	31.104	48.410	3.40	.283	88	12.33
9	14.60	16.483	20	11.5	26.496	44.479	3.43	.286	89	12.45
10		Suction Pipe Off.								1.00

CENTRIFUGAL TEST.

Test No.	Weir Constant.	R.P.M. Pump.	R.P.M. Dyn.	Dyn.		H.P. of Pump.	Q per.sec.	Eff. %	Capacity.
				corrected Scale	H.P.				
1	.6085	974	176	84.7	4.189	2.141	.357	51.0	230000
2	.6060	1000	172	92.2	4.490	2.380	.412	54.3	265000
3	.6056	1030	180	93.2	4.752	2.300	.427	48.4	275000
4	.6105	1010	180	80.2	4.050	1.973	.315	48.6	203000
5	.6110	1044	180	79.2	3.996	1.935	.304	48.5	196000
6	.6083	1010	174	86.2	4.225	2.193	.360	51.8	292000
7	.6067	1030	174	90.2	4.437	2.328	.405	52.5	260000
8	.6063	984	180	93.7	4.768	2.221	.406	46.4	262000
9	.6060	990	174	95.2	4.698	2.068	.412	44.1	265000
10		1048	180	7.2	0.317	0.000		00.	

TRIPLEX FRICTION LOADS.

Dyn. Scales	Dyn. Scales Correct.	Dyn. Revo.	Strokes.	H.P	
111	14.2	180	80	.648	Suction off.
104	7.2	180	80	.324	Plungers #2 + #3 off.
108	11.2	180	80	.504	Plunger #3 off.
98.25	1.5	180	80	.058	Belt and Countershaft.
99.5	2.7	180	80	.123	Mainshaft.
96.75 = Zero.					

WEIR CALIBRATION.

Time	Hook Gauge.	Cu. Ft.	
10.30	30.18"	390.5	Leakage = .05 cu ft per min.
10.40	30.15	391.0	
10.49	28.92	412.5	running over weir.
10.54	27.71	432.5	
10.59	26.54	454.0	Head on weir = .98 inches, the
11.04	25.39	474.5	quantity = 4.2 cu ft. per. min.
11.10	24.00	497.5	
11.15	22.90	523.8	
11.19	21.98	540.0	

Centrifugal.
Pulley – 6" dia. $6\frac{1}{2}$" face
2" Suction
$1\frac{1}{2}$" Delivery

Triplex.
Pulley – 16" dia. 3" face.
2" Suction Pinion – 18 teeth.
$1\frac{1}{2}$" Delivery Gear – 90 " .

R.P.M. of Line Shaft = 220

TRIPLEX TEST.

Time.	Counter.	Vac. Inches	Vac. Ft.	Correct Pres.	Pres. Ft	Total Head	Dyn.	R.P.M. Dyn.	R.P.M. Pump.
				TEST No. 1.					
1-40	33569	1.5	1.69	5	11.52	14.21	116	178	80
2-00	35181	1.5	1.69	5	11.52	14.21	116	178	80
2-10	35993	1.5	1.69	5	11.52	14.21	116	178	80
				TEST No. 2					
2-25	36600	1.3	1.47	19	43.776	46.246	118	178	80
2-40	37798	1.3	1.47	19	43.776	46.246	118	178	80
2-55	39000	1.3	1.47	19	43.776	46.246	118	178	80
				TEST No. 3.					
3-05	39807	1.4	1.58	35	80.64	83.220	125	178	80
3-20	41000	1.5	1.69	35	80.64	83.330	125	178	80
3-35	42199	1.4	1.58	35	80.64	83.220	125	178	80
				TEST No. 4					
3-45	42997	1.5	1.69	50	115.20	117.89	131	178	80
4-00	44192	1.5	1.69	50	115.20	117.89	131	178	80
4-15	45385	1.5	1.69	50	115.20	117.89	131	178	80
				TEST No. 5					
4-21	45865	1.4	1.58	80	184.32	186.90	144	178	80
4-36	47040	1.5	1.69	80	184.32	187.01	144	178	80
4-51	48228	1.4	1.58	80	184.32	186.90	144	178	80
				TEST No. 6					
4-16	49036	1.5	1.69	101	232.7	235.4	150	178	80
4-26	49832	1.5	1.69	101	232.7	235.4	150	178	80
4-37	50707	1.5	1.69	101	232.7	235.4	150	178	80
				TEST No. 7.					
4-51	51089	1.3	1.47	122	281.1	283.7	156	178	80
5-01	51884	1.4	1.58	122	281.1	283.8	156	178	80
5-06	52280	1.4	1.58	122	281.1	283.8	156	178	80

Triplex Test.

Head on Weir. Ft.	Temp.	Q. per min.	Delivered H.P.	Dyn. H.P.	Indicated H.P.	% Eff. Mech	Eff. Pump.	Capacity.	Card. No.
				Test No. 1					
.081	83	4.2							1
.081	83	4.2	.112	.89	.119	13.4	94.1	45100	2
.081	83	4.2							
				Test No. 2					
.081	83	4.2							3
.081	83	4.2	.366	.981	.394	40.5	93.0	45100	4
.081	83	4.2							
				Test No. 3					
.081	83	4.2							5
.081	83	4.2	.657	1.31	.732	56.0	89.5	45100	6
.081	83	4.2							7
				Test No. 4					
.081	83	4.2							8
.081	83	4.2	.934	1.602	1.04	64.8	89.8	45100	9
.081	83	4.2							10
				Test No. 5					
.081	83	4.2							11
.081	83	4.2	1.480	2.264	1.58	69.5	93.8	45100	12
.081	83	4.2							13
				Test No. 6					
.081	83	4.2							14
.081	83	4.2	1.855	2.577	1.97	76.4	94.1	45100	15
.081	83	4.2							
				Test No. 7					
.081	83	4.2							16
.081	83	4.2	2.245	2.890	2.38	82.4	94.1	45100	17
.081	83	4.2							

Table for Dynamometer Curve.

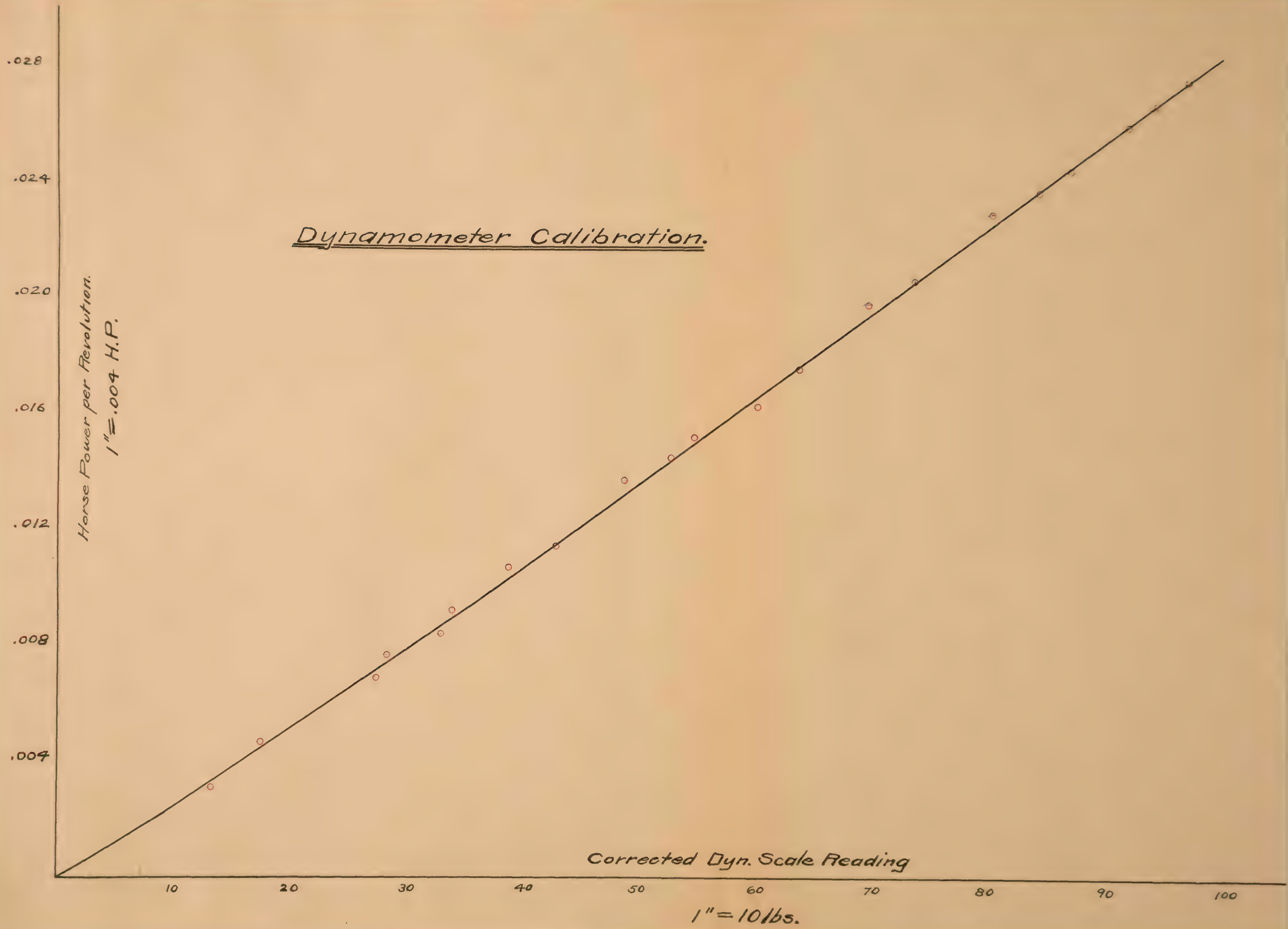
Test May 4 (#3)

H.P. per Revolution.	Dyn. Scale.
.00685	27.2
.00837	32.7
.01142	42.7
.01446	52.7
.01750	63.7
.02055	73.7
.02359	84.2
.02664	94.2

Curve on next page.

Test May 6, 1901.

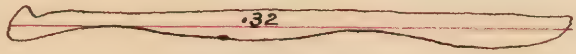
H.P. per Revolution.	Dyn. Scale.
.00304	13.2
.00457	17.2
.00761	28.2
.01065	38.7
.01370	48.7
.01619	60.2
.01979	69.7
.02283	80.2
.02435	86.7
.02740	96.7
.02588	91.7
.01522	54.7
.00913	33.7



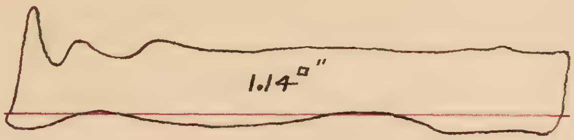
Dynamometer Calibration.
Horse Power per Revolution.
1" = .004 H.P.
.028
.024
.020
.016
.012
.008
.004
Corrected Dyn. Scale Reading
10
20
30
40
50
60
70
80
90
100
1" = 10 lbs.

Sample Cards Cyl. No. 1.

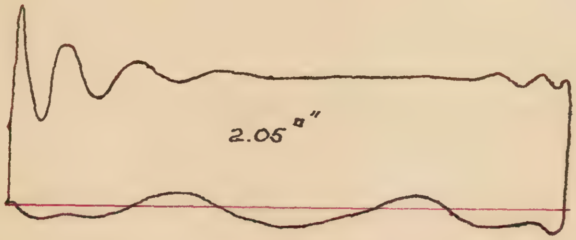
SPRING SCALE—52.3



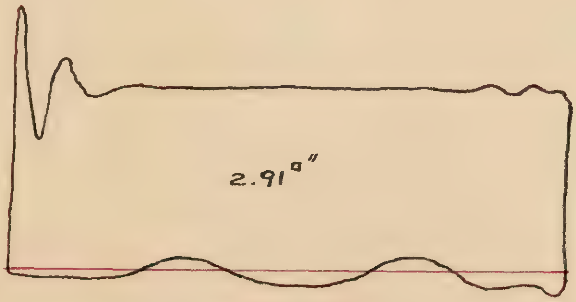
LOAD No. 1



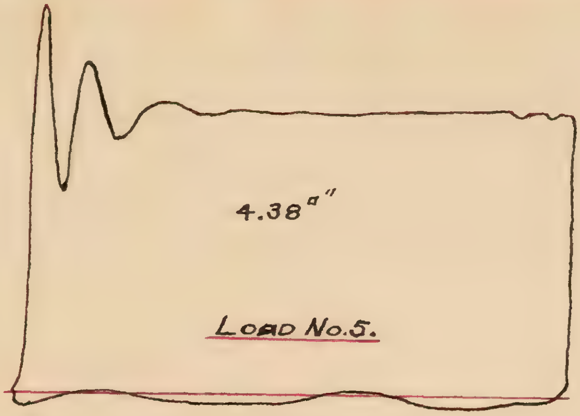
LOAD No. 2.

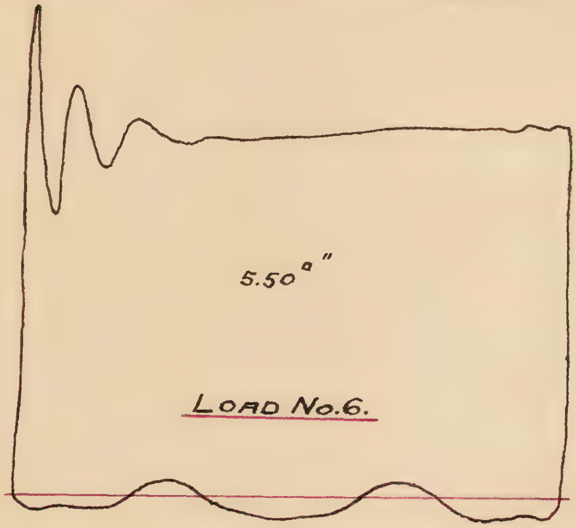


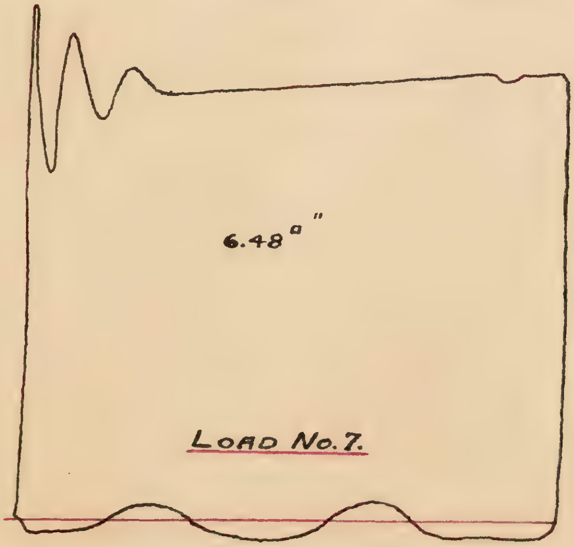
LOAD No. 3.



LOAD No. 4.


4.38□"
LOAD No.5.


5.50□"
LOAD No.6.


6.48□"
LOAD No.7.

Card. No.	Cylinder No. 1.				Cylinder No. 2				Cylinder No. 3				No. of Strokes.
	Area	average m.e.p.	Correct M.E.P.	Stroke.	Area.	average M.E.P.	Correct M.E.P.	Stroke	Area	Average M.E.P	Correct M.E.P.	area.	
1	.35				.4				.39				80
2	.32	5.75	6.02	3"	.4	5.65	6.00	3"	.37	5.49	5.9	3"	80
3	1.14				1.32				1.28				80
4	1.12	19.2	20.1	3"	1.29	18.35	19.6	3"	1.28	18.4	19.8	3"	80
5	2.05				2.47				2.41				80
6	2.07				2.36				2.35				80
7	2.13	35.26	37.0	3"	2.41	34.07	36.2	3"	2.42	34.6	37.1	3"	80
8	2.91				3.45				3.37				80
9	2.94				3.47				3.40				80
10	2.86	49.73	52.0	3"	3.46	48.46	51.5	3"	3.39	49.15	52.9	3"	80
11	4.09				5.35				5.29				80
12	4.38				5.27				5.27				80
13	—	72.8	76.2	3"	5.17	74.26	78.8	3"	5.29	77.	82.6	3"	80
14	5.52				5.42				2.85				80
15	5.50	94.7	99.0	3"	5.39	94.8	99.3	3"	2.75	99.3	99.3	3"	80
16	6.50				6.51				3.46				80
17	6.48	112	117.	3"	6.47	114	119	3"	3.48	123.	123.	3"	80

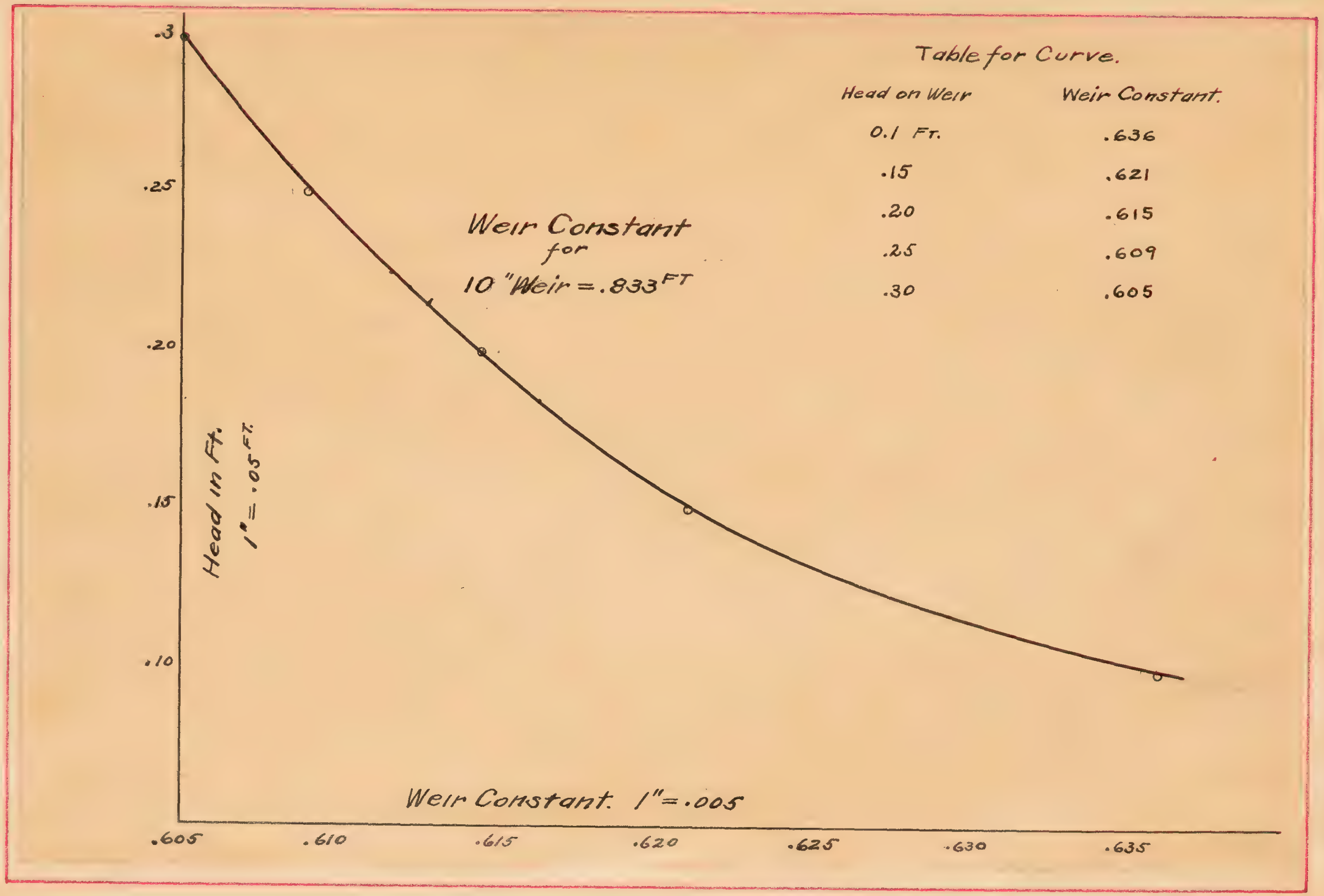


Table for Curve.
Head on Weir
Weir Constant.
0.1 Ft.
.636
.15
.621
.20
.615
.25
.609
.30
.605
Weir Constant
for
10" Weir = .833 FT
Head in Ft.
1" = .05 FT.
.3
.25
.20
.15
.10
Weir Constant. 1" = .005
.605
.610
.615
.620
.625
.630
.635

Calibration of Gauge 189439.
used on Triplex.

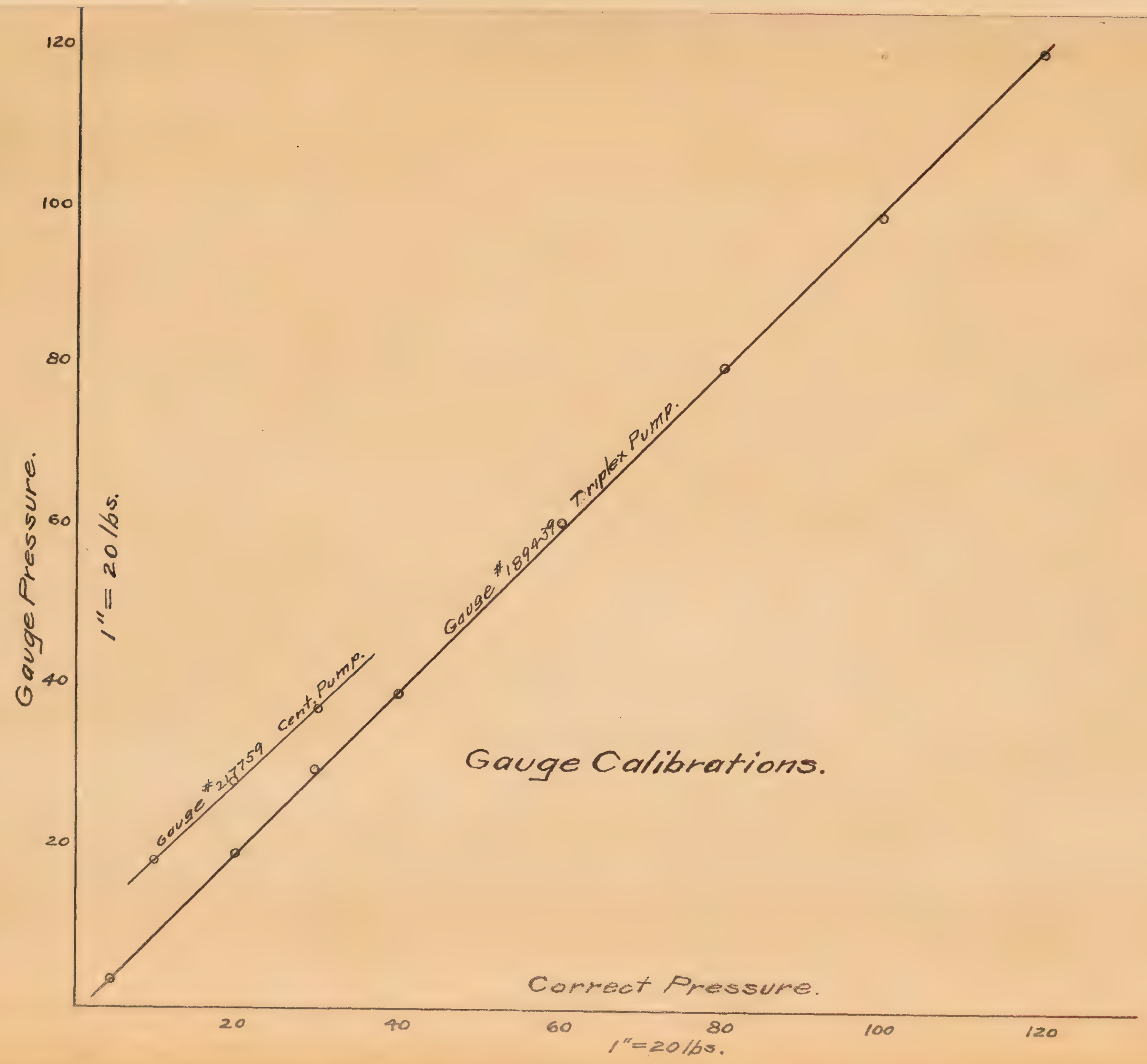
True.	Asc.	Desc.	Mean.
10	9	10.5	9.8
20	19	20.2	19.6
30	29.8	30.2	30.0
40	38.	40.6	39.3
50	48.	50.8	49.4
60	59.6	61.4	60.5
70	68.9	71.9	70.4
80	79.4	81.0	80.2
90	90.0	91.0	91.5
100	99.4	100.2	99.0
110	108.8	110.4	99.8
120	120.0	120.0	120.0

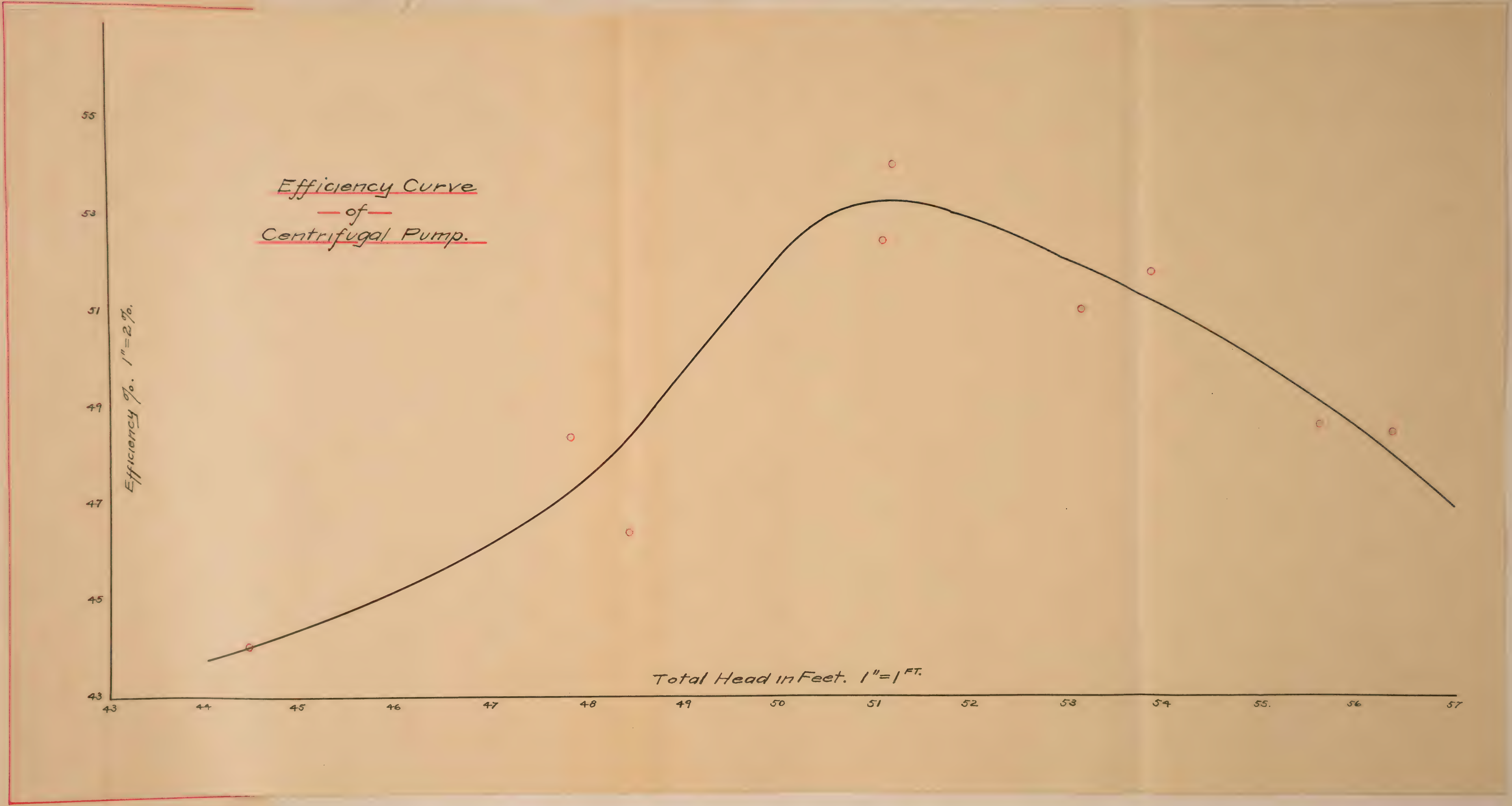
curve on next page.

Calibration of Gauge 217759.
used on Centrifugal.

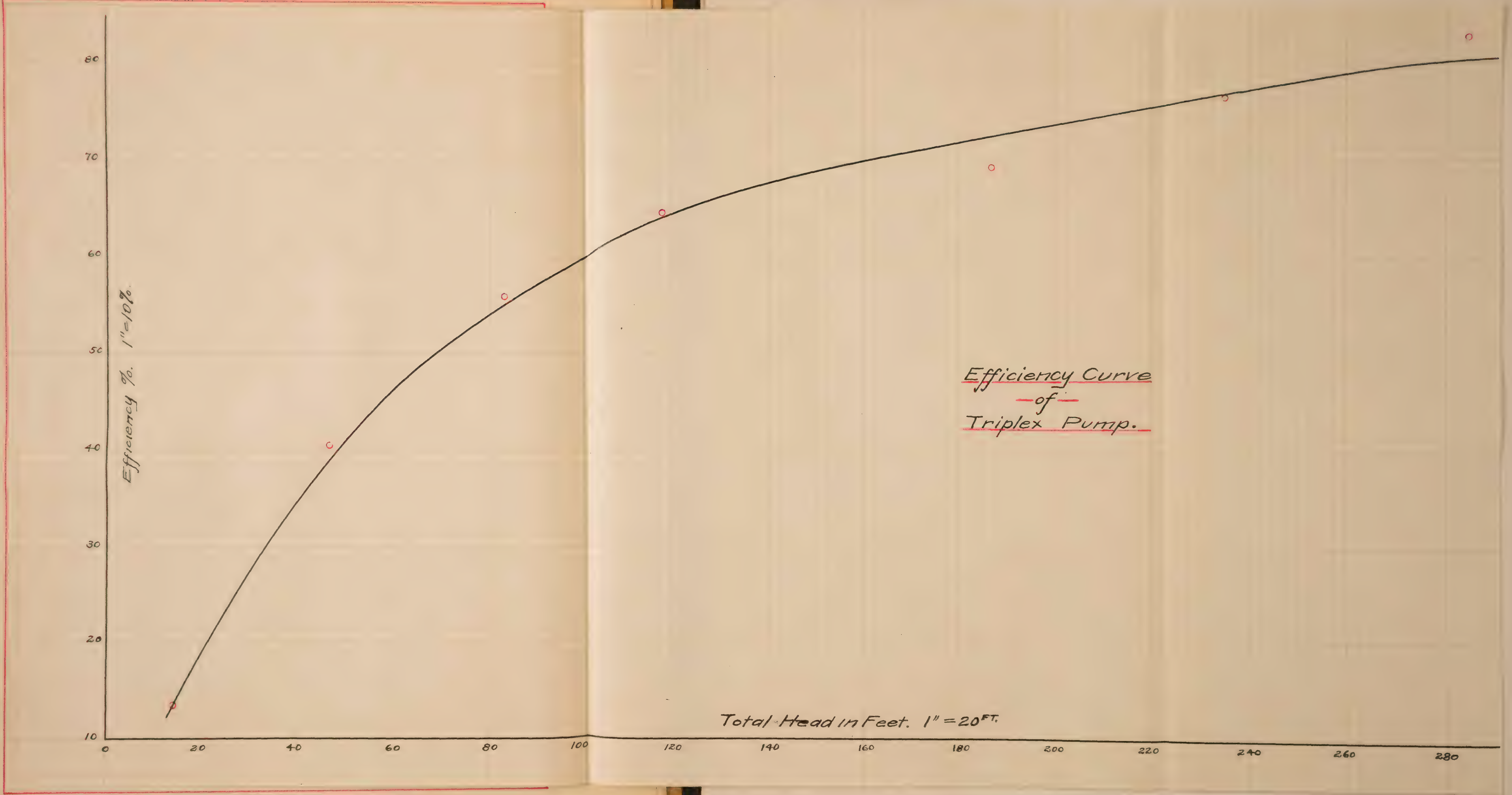
True	Asc.	Desc.	Mean.
10	19	18	18.5
15	23	23	23
20	28	28	28
25	33	33	33
30	37.5	37.5	37.5

Curve on next page.


Gauge Calibrations.
Gauge Pressure.
1" = 20 lbs.
Correct Pressure.
1"=20 lbs.
Gauge #189439 Triplex Pump.
Gauge #217759 Cent. Pump.
20
40
60
80
100
120
20
40
60
80
100
120



Efficiency Curve
— of —
Centrifugal Pump.
Efficiency %. 1"=2%.
Total Head in Feet. 1"=1 FT.
55
53
51
49
47
45
43
43
44
45
46
47
48
49
50
51
52
53
54
55.
56
57



Efficiency Curve
—of—
Triplex Pump.
Efficiency %. 1"=10%.
Total Head in Feet. 1" = 20 FT.
80
70
60
50
40
30
20
10
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280